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Centrifugal-Pump Performance as Affected by Design Features

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This paper presents some of the results of a study of Grand Coulee pumping-plant characteristics. The research program was conducted for the Bureau of Reclamation by the California Institute of Technology in its hydraulic machinery laboratory (1)[2] and has been in progress since January, 1938. While the principal object was to determine the operating features for pumping units to be installed at the Grand Coulee project, the results obtained are somewhat more generally applicable than might be expected. It is believed by the author that investigations of a somewhat similar nature offer the most reliable means for securing the characteristics desired in hydraulic units, both pump and turbine, for practically any given set of conditions.

Need for Information

The need for a thorough study of the Grand Coulee pumping plant arises basically from the tremendous size of the units proposed, i.e., 1600 cfs capacity with a motor of approximately 65,000 hp for each pump. Full knowledge of the pump characteristics is required, due to the great range of operating head, from 295 to 367 ft, which is caused by the variation of the inlet head, from + 80 ft to + 5 ft. Also, the probable operating cycle makes it desirable to have as high a capacity as possible when operating against the high head. In addition to the matter of the proper relationship between the capacity and head over the operating range, the following items were considered to be important for satisfactory pump operation:

(*a*) Freedom from cavitation over the entire operating range
(*b*) Low radial forces due to hydraulic unbalance.
(*c*) Freedom from unstable regions within the operating range.
(*d*) Constant-speed operation.
(*e*) Satisfactory transient performance which will permit simple shutdown procedure.
(*f*) Suitable characteristics when operating as a turbine to provide the possibility of utilizing units for peak-load power development.

Furthermore, to obtain the lowest-cost unit, including the motor, it was necessary to determine the maximum permissible operating speed for which units could be obtained that could also satisfy the foregoing requirements.

Model and Prototype Pumps

The pumps contemplated for installation at Grand Coulee are unprecedented in size and power requirement. They are to be installed vertically and will be of the single-stage single-suction type. Each unit is expected to have a capacity of about 1600 cfs. The normal head against which it is to deliver is about 295 ft. Approximately 65,000 hp will be required. It is estimated that the pump for this duty will have a discharge nozzle of 8 to 10 ft diam, an impeller of from 12 to 17 ft diam, with an eye dimension of from 6 to 10 ft. The width of the impeller at discharge will be in the neighborhood of 20 to 36 in. The speed range is from 150 to 200 rpm or possibly slightly higher.

Although the present study is not a "model" study, but rather an investigation of the possible characteristics of the machines, the units tested in the laboratory may be thought of as models in order to visualize a size comparison. On this basis the model ratio would range from $12^1/_2$ to 15. The studies were all made at or near the full prototype head. The capacities varied from 7 to 10 cfs at the operating point. The horsepower requirements fell within the range of 290 to 400. All of the units had maximum efficiencies in the vicinity of 90 per cent. The discharge-nozzle diameters were 8 in. The impellers varied from $12^1/_4$ to $14^1/_2$ in. diam, with eyes of from 6 to 8 in. and with discharge widths of from $1^1/_2$ to $2^1/_2$ in. Testing speeds fell between 2100 and 2600 rpm.

It will thus be realized that these test pumps are comparatively large machines, therefore, accurate passages and vane angles may be expected. Furthermore, the large size and high efficiency of these units permit drawing direct conclusions concerning the performance of prototypes. To reduce the number of variables, several cases were designed to operate with the same impeller, thus making it possible to ascertain clearly the characteristic-performance differences between such case types as single-volute, double-volute, and fixed-vane-diffusor constructions.

Presentation of Data

In order to make the results from the different units directly comparable, the characteristic curves have been plotted on a percentage basis. The normal operating head at Grand Coulee is 295 ft. This has been taken as 100 per cent. The capacity at this head is therefore designated as 100 per cent. The maximum efficiency of each unit has been used as the reference value for that unit, and has been plotted as 100 per cent. It should be noted that the maximum-efficiency point will not coincide necessarily with the 100 per cent capacity and head point. Whenever plotted, torques and horsepowers have had, as a 100 per cent reference, the corresponding values at 100 per cent capacity and head. For example, since the prototype-head range is from 295 to 367 ft, this system gives an operating-head range of from 100 to 125 per cent.

Comparison of Normal Operating Characteristics

Capacity-Head and Efficiency Characteristics. During the course of this program, several series of experiments were made in which a single impeller was tested in two or three different cases. In order to establish a basis for the discussion of the results, a brief résumé of the respective functions of the impeller and the case of a centrifugal pump seems desirable.

The impeller adds energy to the fluid flowing through it. At the discharge from the impeller, this added energy is in two forms: (*a*) an increase in pressure, and (*b*) an increase in velocity. The case has two functions: (*a*) to collect the fluid as it discharges

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[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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around the impeller periphery, and (*b*) to transform a large part of the velocity into pressure with as little loss of energy as possible.

If an impeller could be tested alone under such conditions that, for all rates of flow, the discharge would be uniform around the periphery, its basic operating characteristics would be determined. A perfect case would be one which would have no losses over the entire operating range; therefore the combination of the impeller, operating in such a case, would have the identical performance characteristics which were obtained from the impeller operating alone. Since no real case is without losses and, furthermore, since no real case is equally efficient over the entire operating range, the performance of the unit as a whole is always lower than that of the impeller alone. The deviation will be least in the zone in which the case characteristics match the impeller

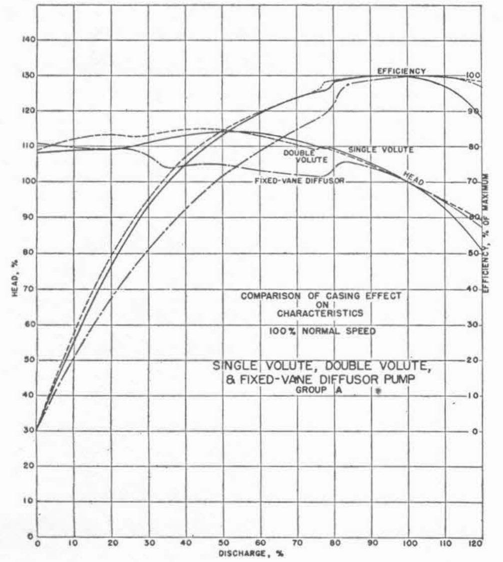

FIG. 1 COMPARISON OF CASING EFFECT ON PUMP CHARACTERISTICS; GROUP A, 100 PER CENT NORMAL SPEED

characteristics to best advantage, and will increase on both sides of this zone. For a good pump, the case must match the impeller within the high-efficiency zone of the latter.

The case will affect the over-all performance of the pump in two ways: (*a*) through energy losses in the case itself, and (*b*) in additional energy losses induced in the impeller. Fundamentally, the case can affect the impeller performance only in one way, i.e., by varying the pressure distribution around the periphery of the impeller and thus producing nonuniform discharge. If the discharge is not uniform from all parts of the impeller periphery, it follows that there must be pulsating flow in the impeller passages, nonuniform entrance conditions at the eye, and presumably increased losses, both in the impeller and in the case. In this simplified picture, the secondary effects of the impeller shrouds, the circulation existing between them, and the casing walls and leakage losses to the suction sides are disregarded.

From this discussion, it will be realized that a comparison of the characteristics of different units, made up of various types of cases operating with the same impeller, resolves itself into a comparison of the relative matching of these cases to the impeller and of casing losses, both intrinsic and induced in the impeller. Fig. 1 shows such a comparison for a series of units designated as group A. The first unit was designed as a single-volute pump to operate at a prototype speed of 150 rpm. The double-volute case was then constructed, using the same design methods. It was anticipated that, if everything worked out satisfactorily, the performance of the double-volute pump would be the same as that of the single-volute unit. The fixed-vane-diffusor case was designed around the same impeller.

If the curves for the single- and double-volute cases are compared, a striking difference is observed in the high-capacity region. The head curve for the double-volute case does not fall off as rapidly as that for the single-volute pump, and the efficiency also remains higher. The same is true to a lesser extent in the low-capacity region. However, the maximum efficiency is about the same. Since these maximum efficiencies are high, both cases are very satisfactory in the region of the design point, but the double-volute case apparently matches the impeller characteristics better in the low- and high-capacity regions. It must be remembered that, because of the two passages, the double-volute case has a lower effective hydraulic radius and, therefore, a higher skin-friction loss. For this reason, the wide region of high efficiency is all the more surprising.

The fixed-vane-diffusor case, operating with the same impeller, shows the same high maximum efficiency observed in the other two cases. However, the characteristic curves are quite different in shape. The maximum-efficiency point comes at a somewhat higher capacity for the diffusor case, and this maximum efficiency is not sustained over as wide a region. This is reflected in the head-capacity curve. It will be noticed on both sides of the design point that the diffusor-case head curve lies

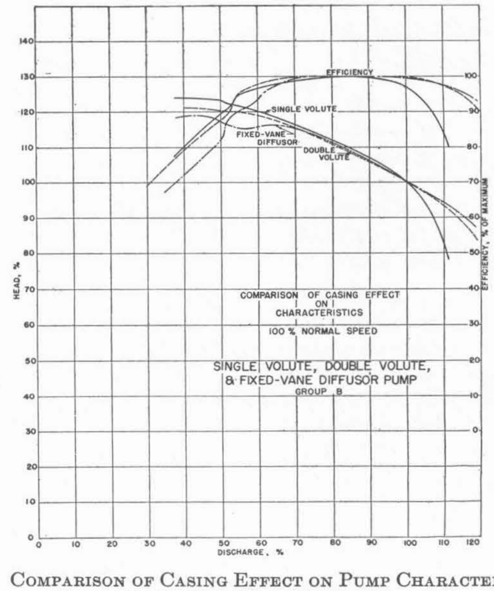

FIG. 2 COMPARISON OF CASING EFFECT ON PUMP CHARACTERISTICS; GROUP B, 100 PER CENT NORMAL SPEED

under that of the double-volute curve. In the low-capacity region, i.e., from zero up to 75 per cent, the efficiency of the diffusor is markedly lower than that of the other two cases. This is probably the result of the discrepancy between the angle of the fixed guide vanes and that of the flow leaving the impeller under these conditions.

Fig. 2 shows the same comparison for an entirely different set of cases, working with another impeller. This series of units, group B, was designed for a prototype speed of 180 rpm in comparison with the 150-rpm speed of group A. Since the head and capacity are fixed, this 20 per cent increase in operating speed results in a 20 per cent increase in the specific speed as well, which corresponds approximately to a 16 per cent decrease in the diameters of the impeller and the base circle of the case.

The relative performances of the single- and double-volute cases are practically the same as those observed for group A, except that the single-volute pump shows its peak efficiency between 80 and 90 per cent of design discharge. This indicates that the case is too small for the specified conditions. The result is that, at the normal operating point, the efficiency is only about 97 per cent of the maximum. This accounts for the fact that its head-capacity curve is apparently above those for the double-volute

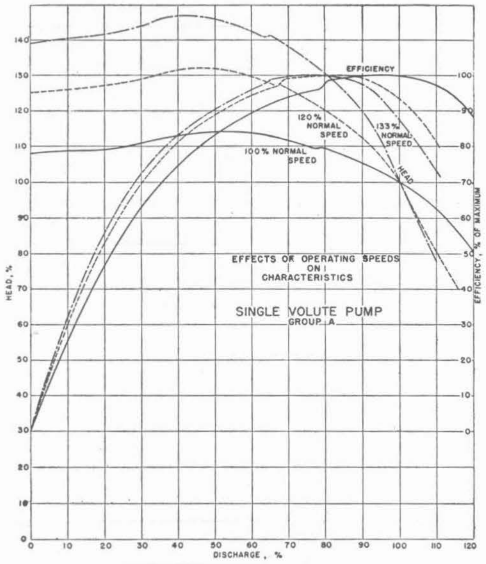

FIG. 3 EFFECTS OF OPERATING SPEEDS ON CHARACTERISTICS; SINGLE-VOLUTE PUMP

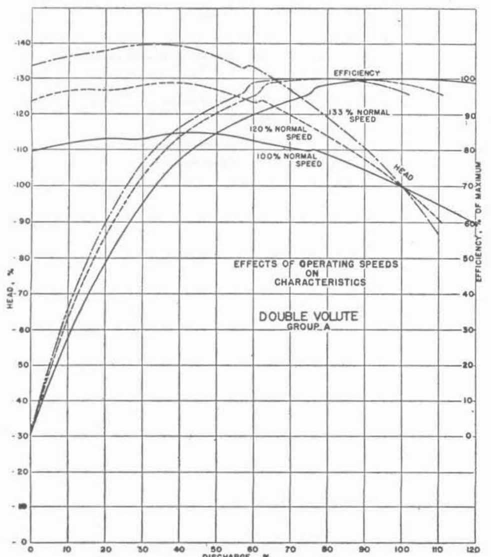

FIG. 4 EFFECTS OF OPERATING SPEEDS ON CHARACTERISTICS; DOUBLE-VOLUTE PUMP

type and fixed-vane-diffusor case, i.e., the steepness is obtained by sacrificing efficiency. The double-volute case again shows a surprisingly wide range of high-efficiency operation but, in this series, the diffusor case nearly duplicates its performance. However, the sharp drop in efficiency for the low-capacity region is again observed to be a diffusor-case characteristic.

Choice of Operating Speed. A given pump is designed to operate at a definite specific speed. In general, test results show that the unit has its maximum efficiency at this condition. However, if the performance characteristics show a reasonably broad zone of high efficiency, it may be possible to secure a better agreement between the pump characteristics and the field requirements if a different operating speed is chosen. The effect of the choice of operating speed may be observed in Figs. 3, 4, and 5. Fig. 3 shows the performance of the single-volute unit of group A operating at speeds of 100, 120, and 133 per cent of the design value. Fig. 4 presents the corresponding performance of the double-volute case, and Fig. 5 that of the fixed-vane pump.

All three units show the same trend, i.e., a marked steepening of the head-capacity characteristics with increase in operating speed. A closer examination of the three sets of curves shows that there are apparently two causes for this increase in steepness, (*a*) an increase due to the normal increase in the steepness

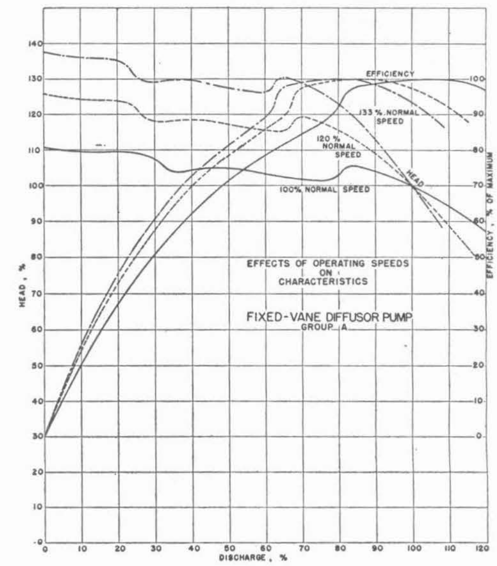

FIG. 5 EFFECTS OF OPERATING SPEEDS ON CHARACTERISTICS; FIXED-VANE-DIFFUSOR PUMP

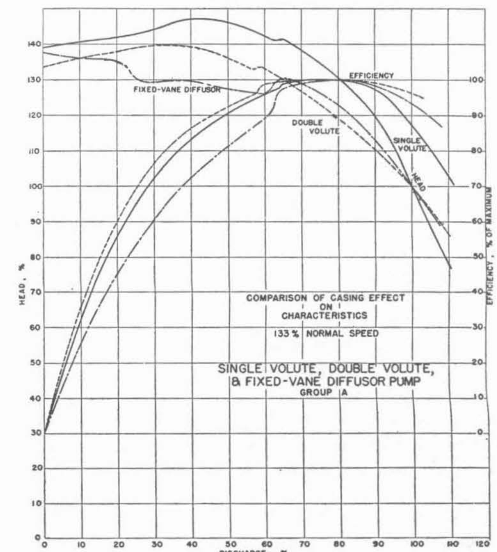

FIG. 6 COMPARISON OF CASING EFFECT ON PUMP CHARACTERISTICS; GROUP A, 133 PER CENT NORMAL SPEED

of the impeller characteristic, as the capacity is increased, and (*b*) an increase in steepness due to a decrease in the efficiency of the case. Fig. 4 illustrates the effect of the former. It will be noted that, in the region of from 80 to 100 per cent discharge, the efficiency is high for all speeds, in fact, the lowest value is 96.5 per cent of the maximum. Thus, the change in steepness for this machine must be due largely to the shape of the impeller characteristics. The fixed-vane pump, Fig. 5, shows a larger variation in steepness, but the efficiency drops to 92 per cent in the same capacity range. Likewise, the single-volute pump, Fig. 3, shows an even greater variation in steepness, but the efficiency goes down to about 87 per cent of the maximum.

The difference between these three cases, when operated at the higher speed, is shown very clearly in Fig. 6. Here, the performance characteristics for the same three units, presented in Fig. 1, are plotted for a speed of 133 per cent of normal. At the design speed of Fig. 1, the head-capacity characteristic of each of the three cases shows about the same slope in the vicinity of the operating point. At the 33 per cent overspeed, however, the difference in steepness is quite marked.

From these comparisons, it would seem that, in view of the factors so far considered, the steepness of the head-capacity characteristics can be varied appreciably by choosing the speed at which the pump is to operate. If the choice is limited to speeds within the high-efficiency range, slight loss accompanies the variation. The double-volute case offers the widest possibilities within these limits because of its broad zone of high-efficiency performance. If steeper characteristics than those corresponding to the basic impeller performance are desired, they can be obtained only through sacrifice of efficiency. It should be remembered, however, that in this investigation no attempt has been made to explore fully the possibility of varying the impeller characteristics themselves.

Minor Operating Features

Hydraulic Balance and Radial Thrust. In the section, "Comparison of Normal Operating Characteristics," it was stated that the case can affect the impeller performance only by varying the pressure distribution around the periphery of the impeller and thus producing nonuniform discharge. Since this is an important feature for pump operation, it was thought desirable to make some experimental determinations of the pressure variation in the volute for the different types of cases. Consequently, piezometer connections were installed in the various cases—they were at constant radius. The piezometers for each case were spaced around a circle the diameter of which was slightly greater than the impeller and they covered the full 360 deg. Thus, the readings from them give a good picture of the pressure distribution around the impeller discharge.

Figs. 7, 8, and 9 show these measurements for the three cases of group B. The ordinates of all three curves are the static pressure at the piezometer connections, expressed in a percentage of the normal head produced by the pump. If the measurements for the single-volute pump, Fig. 7, are studied, it will be seen that the pressure distribution is reasonably uniform in the vicinity of the normal capacity; in fact, the most uniform distributions of those shown seem to be for the 93 per cent capacity. A glance at Fig. 1, shows that this is about the point of maximum efficiency. For higher and lower capacities, the pressure distribution is far from uniform and must affect the impeller discharge appreciably.

Fig. 8 shows that, for the double-volute pump, conditions are quite similar except that, of course, there are two pressure cycles in the 360 deg of the case. It will be noted here, however, that the range of pressure variation is considerably lower than in the corresponding single-volute case, although the basic design factors are similar.

Fig. 9 shows that the fixed-vane-diffusor pump has an even lower range of pressure variation. It should be remembered that these pressures are taken at a diameter corresponding to that of the impeller, i.e., at the inner side of the guide vanes. It will be seen that the pressure distribution is still nonsymmetrical. This is presumably due to the effect of the single volute on the outside of the guide vanes proper.

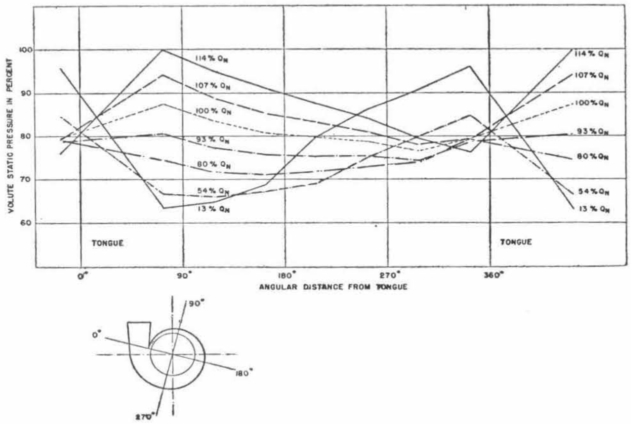

Fig. 7 Static-Pressure Distribution; Single-Volute Pump, Group B

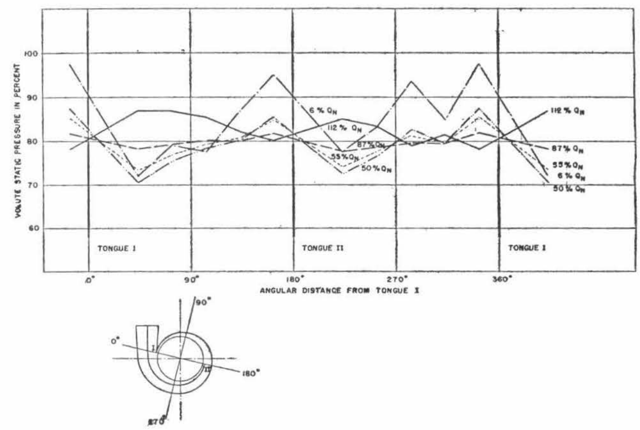

Fig. 8 Static-Pressure Distribution; Double-Volute Pump, Group B

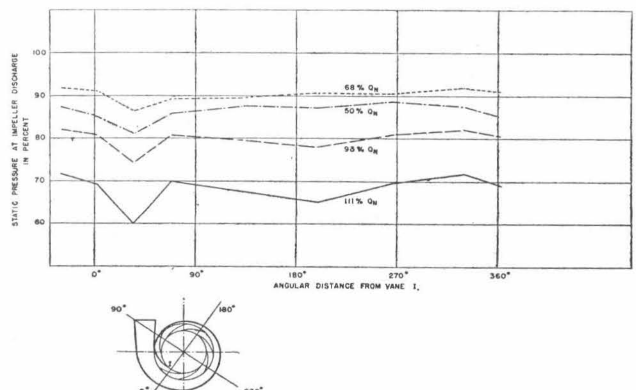

Fig. 9 Static-Pressure Distribution; Fixed-Vane-Diffusor Pump, Group B

Although this pressure variation must have a very marked effect upon the hydraulic performance of the unit, from an operating point of view, there is an even more direct result. A nonuniform pressure distribution such as, for example, the one shown in Fig. 7, for 114 per cent Q_n, indicates that there is a resultant radial thrust upon the impeller. This force must be taken care of in the mechanical design of bearings, case, and shaft, and may well be the controlling factor in the choice of shaft diameter and other important details. Failure to recognize this factor may result in mechanical contact of the wearing-ring surfaces and rapid deterioration of the equipment.

If the pressure-distribution diagrams for the double-volute pump, shown in Fig. 8, are integrated over the 360 deg, it will be found that the resultant radial force is small, since the effect of each of the two volutes nearly cancels the other. This is apparently true for all capacities and represents a distinct advantage of this type of construction. The resultant radial thrust upon the impeller of the diffusor pump, Fig. 9, is much lower than for the single-volute, but is somewhat higher than that of the double-volute. However, it should present no serious design problem, since it is not large.

It should be noted again that the radial unbalance of the fixed-vane-diffusor pump is due to the same cause that produced it in the single-volute pumps, i.e., the presence of the single volute itself. The main reason that the variation in pressure distribution and the resultant thrust are so much lower with the diffusor pump is that the flow is discharged into the volute at a much lower velocity than it is in the case of a single volute. If a fixed-vane case were designed, in which the vanes were used only as stay bolts and not as diffusors, high resultant radial forces should be expected. The importance of the investigation of these radial forces is illustrated by the fact that, for a good single-volute prototype, the unbalanced thrust is of the order of 50 tons. This would make illusory the feature of bearing-load elimination, commonly attributed to the vertical design.

Instability. Figs. 1 and 2 show that there are discontinuities in the head-capacity curves for all six cases. Such discontinuities appear to be characteristic of centrifugal-pump performance and are practically always found whenever tests of sufficient accuracy and detail are made. These discontinuities apparently are the result of a change in the flow from one regimen to another. For different design conditions, it seems that this change in flow can be localized either in the impeller or in the case. In addition, if the change is large enough in the impeller, it may also produce a significant change in the flow in the case. These flow discontinuities produce unstable ranges in the pump performance and, therefore, good practice indicates that the operating zone should not approach them too closely. For example, in the present study, one criterion tentatively proposed is that the maximum operating head should be at least 10 ft (3.5 per cent) below the break in the curve, as it is approached from the high-capacity side. This appears to be a quite satisfactory margin of safety for units having a reasonably small change in head at the discontinuity point, but may be somewhat inadequate for pumps having discontinuities as large as that shown by the fixed-vane diffusor of group A. For such pumps, it would seem advisable to restrict the maximum operating head to 1 or 2 per cent lower than the lowest value at the discontinuity region.

It is interesting to consider that significant information can be obtained by comparing the discontinuity regions, as shown by the capacity-head curves, with the torque or horsepower curves for the same conditions. If the flow regimen changes within the impeller passages, there will be a corresponding difference in the amount of angular momentum imparted to the fluid and this, in turn, will be apparent on the torque and horsepower curves. Thus, it may be concluded that, if a discontinuity in the head-capacity curve is reflected in the torque curve, the change in flow at least originates in the impeller. Conversely, if a discontinuity in the head-capacity curve is not accompanied by a similar break in the torque or horsepower curves, the change in the flow probably is localized in the casing. Unfortunately, space does not permit the plotting of the torque curves in Figs. 1 to 6, inclusive.

Cavitation Limits

Basic Limit of Eye Design. For each given design of an impeller eye, there is a relationship between capacity and inlet head which defines the beginning of cavitation. This basic limit, of course, assumes that, for all capacities, the flow has a normal velocity profile at the pump inlet; that the flow into the eye is circumferentially uniform; and that there are no tangential-velocity components present before the eye is entered. The difference between the basic characteristics of various eye designs for the same specific speeds will depend upon the abilities of the designers to keep their static pressures up and to eliminate local high-velocity regions in the vicinity of the passage entrances.

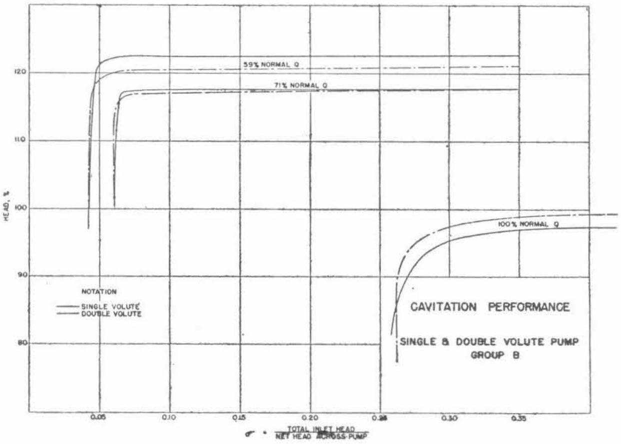

Fig. 10 Cavitation Performance; Single- and Double-Volute Pump, Group B

For a given impeller, however, this basic eye characteristic can be considered as the ideal limit for good performance. In actual operation, it can be modified either by the entrance conditions in the inlet piping approaching the pump, or by the reaction of the case on the inlet flow. The effect of the inlet piping is, of course, an installation problem, and will not be considered here, but the effect of the case is a question of basic pump design.

Effect of Case on Basic Limits. The effect of the case on the impeller characteristics has been discussed previously in the sections, "Comparison of Normal Operating Characteristics" and "Hydraulic Balance." It was seen that, in both high- and low-capacity regions, the case could produce a nonuniform pressure distribution around the impeller discharge. This must result in a pulsating flow in the impeller passages. Cavitation performance under these conditions must differ from that of steady flow. Previous studies at the laboratory (2) have shown that, under some conditions such as quite low capacity, the pressure unbalance on the impeller may be great enough to cause backflow from the case to the eye. Recent investigations also indicate that, in the same low-capacity region, the inlet tips of the impeller vanes may induce a radial-pressure difference sufficient to distort the flow further. It is difficult to separate these two phenomena, but together they seem to explain the "prerotation" which has been observed at times in pump inlets.

In an attempt to ascertain the effect of the various cases on

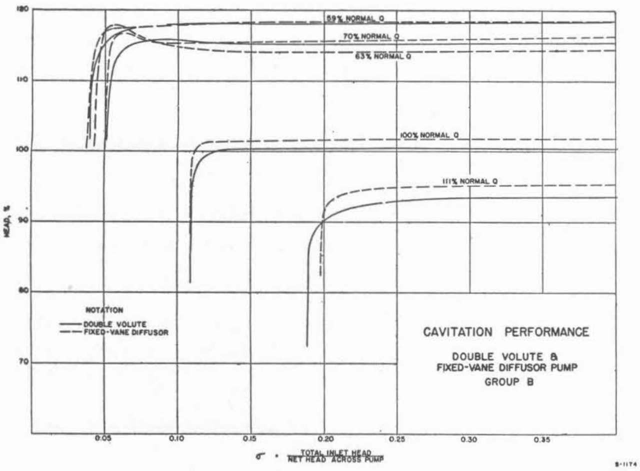

FIG. 11 CAVITATION PERFORMANCE; DOUBLE-VOLUTE CASE AND FIXED-VANE-DIFFUSOR PUMP, GROUP B

the cavitation characteristics of the unit, Figs. 10 and 11 have been prepared. In both figures, the cavitation parameter σ has been plotted against the pump head for a series of constant capacities during which the inlet head was continuously lowered until cavitation was fully developed. Fig. 10 shows the comparative performance of the single- and double-volute cases of group B. It will be noted that the differences are slight, so slight in fact that little significance can be placed upon them. It is unfortunate that no runs are available at very low capacities, since this is the region in which the pressure distribution around the impeller differs widely for the two cases. Fig. 11 compares the double-volute case and the fixed-vane diffusor. These units are also from group B, but the results are not directly comparable to those of Fig. 10, because slightly different impellers were used in the two series of tests. Here, it will be noted that for one capacity the fixed-vane diffusor has a cavitation performance quite different from that shown by all other curves. The head rises rapidly, as σ decreases from 0.12 to 0.06. Since no such behavior is observed for either the single- or double-volute cases, it must be assumed that the fixed-vane-diffusor case is responsible for the difference.

The following logical explanation has been suggested by D. P. Barnes of the Bureau of Reclamation. The capacity at which this deviate behavior occurs is in the region for which the diffusor-vane angles must differ from the calculated discharge angle of the impeller. If cavitation starts in the impeller, it may quite possibly produce a change in the angle at which the flow leaves the impeller. If this angle more nearly coincides with that of the diffusor vanes, then the diffusion should be more effective and, therefore, the pump head should rise. Thus, it is possible that this rising head line on the σ diagram may be an indication of the beginning of cavitation, and hence marks a poorer rather than a better pump performance.

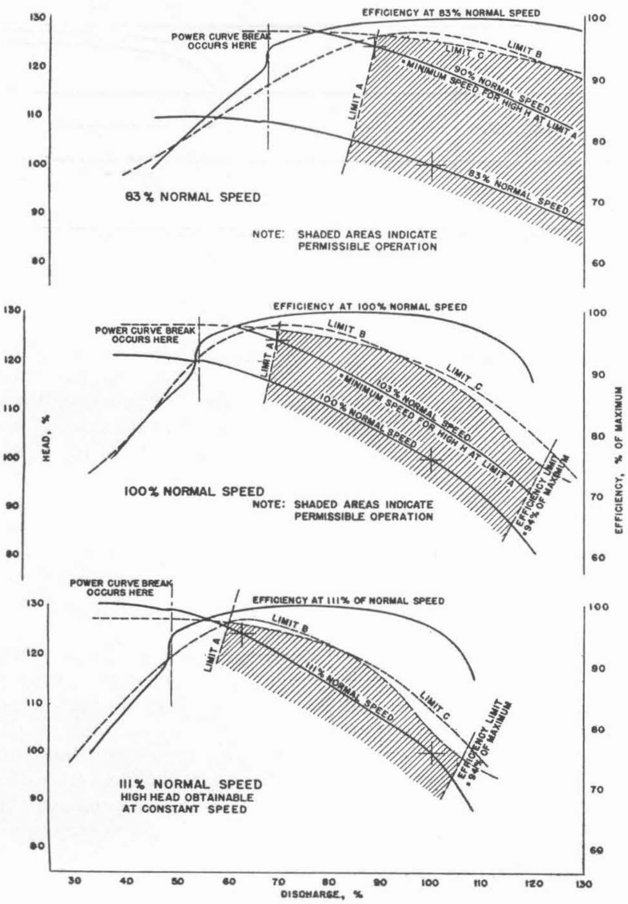

FIG. 12 DIAGRAM SHOWING LIMITATIONS UPON ALLOWABLE OPERATING REGIONS OF CHARACTERISTICS OF DOUBLE-VOLUTE PUMP, GROUP B

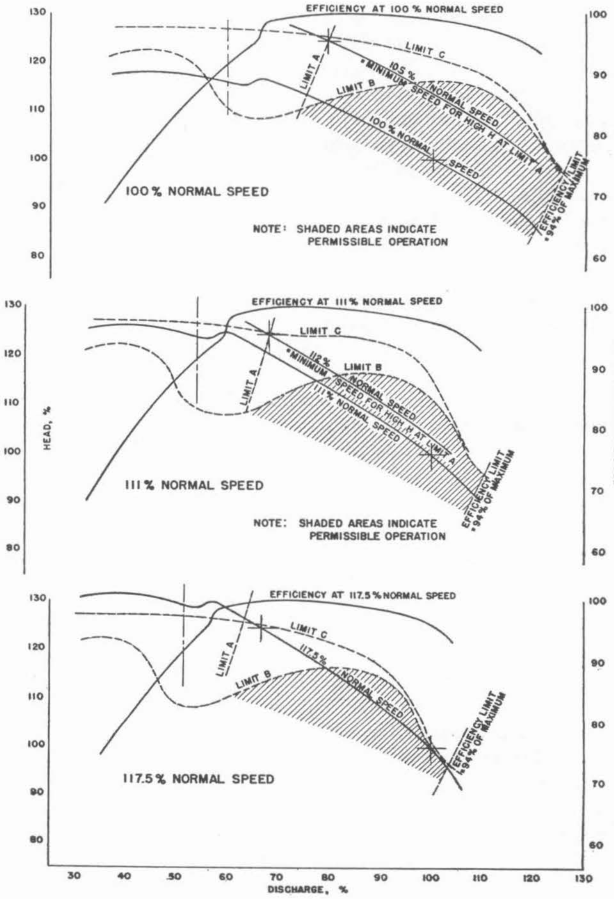

FIG. 13 DIAGRAM SHOWING LIMITATIONS UPON ALLOWABLE OPERATING REGIONS OF CHARACTERISTICS OF FIXED-VANE-DIFFUSOR PUMP, GROUP B

Selection of Operating Region. In an actual pump installation, the physical requirements impose many limitations upon the allowable operating regions of the pump characteristics. For example, the avoidance of discontinuity points has already been discussed. Freedom from cavitation is likewise necessary and this, in turn, is affected by the variation of inlet pressure due to change in reservoir level, etc. Fig. 12 presents a graphical diagram of these limitations as applied to the double-volute case of group B for three possible operating speeds. The limitations imposed are those obtained from a preliminary study of the Grand Coulee conditions. Limit *A* locates the permissible approach to the discontinuity or instability region. Limit *B* bounds the region for freedom from cavitation, as determined by the point at which there is a 0.5 per cent head drop on the σ curve (Fig. 11). Limit *C* bounds another cavitation parameter which is somewhat more complicated but which may be more satisfactory for certain units. The maximum and minimum operating heads are shown by the large crosses. The high-capacity boundary of the zone of permissible operation is arbitrarily defined by the condition that the efficiency has dropped to 94 per cent of the maximum value. The zone of permissible operation is indicated by the cross-hatched area and, within this zone, all of the criteria are met. This very useful type of presentation has been developed by D. P. Barnes.

It will be noted that, when the unit is operated at a speed of 83 per cent of the design value, only the low-head high-capacity portion of the required operating region can be covered. To obtain the high head required, an increase in speed to 90 per cent of the design value is necessary but, if this speed variation is permissible, the entire operating region can be covered satisfactorily. Conditions at the design speed are somewhat similar except that, to meet the high head condition, a speed increase to only something over 103 per cent is required. Operation at a speed of 111 per cent, however, permits the entire operating range to be obtained within the zone of permissible operation at constant speed.

Fig. 13 shows a similar diagram for the fixed-vane-diffusor pump of group B. Here, however, it is seen that, over a range of from 100 to 118 per cent of design speed, it is impossible to find any combination of constant- or variable-speed operation which will cover the desired range and yet meet the limitations imposed. It will be noted that in this unit the most serious deviation from limitations is from the "limit *B*" cavitation parameter.

Turbine Operation for Steady and Transient Conditions

One of the characteristic features of a pump installation is that transient conditions are quite commonly encountered under which the pump is called upon to operate as a turbine. Thus, for example, if the pump is operating normally and power should fail, unless there is a check valve in the line, the unit will slow down, reverse, and come up to runaway speed as a turbine, thus passing through the region of pump operation, a region of complete energy dissipation, and through the entire zone of turbine operation. In the design of large pump installations it is, therefore, very important for the plant designer to know the characteristics of the machines over the entire range of operating possibilities, in order that adequate provision may be made for maximum shaft torques, pressure surges, centrifugal forces, etc.

Complete Characteristic Diagrams. One of the first investigations of this complete range of pump operation was made by Kittredge and Thoma (3). It is convenient to present this information on a single diagram (4). Figs. 14 and 15 are two such diagrams for the single- and double-volute pumps, respectively, of group B. It will be noted that families of constant-head, constant-torque, and constant-efficiency lines are plotted against coordinates of capacity and speed. The performance of the unit at any constant speed is given by the intersection of these families of contours with a vertical line passing through the speed chosen.

Turbine Runaway Speed. The runaway speed of the unit, when operating as a turbine, is given by the intersection of the zero-torque line in the turbine region with the head curve corresponding to the pressure across the pump for that particular condition. For short pipe lines of ample proportions, this head is nearly the same as the pumping head since, under these conditions, the friction losses would be quite small. If the runaway speed exceeds the operating speed by a sufficient margin, it may be the controlling factor in the structural design of the impeller. Since the absolute value of this runaway speed is constant for a given unit operating under a given head, its value relative to the operating speed is determined by the choice of the latter. This can easily be seen by referring to Fig. 14. Consider that the normal operating head is represented by the 100 per cent head line. For the Grand Coulee installation, the maximum possible head which can cause turbine operation is about 120 per cent. The 120 per cent head line intersects the zero-torque line in the turbine zone at a negative speed of about 135 per cent. With a runaway speed of 35 per cent above that of normal operation, the impeller stresses may become quite serious. However, if it were decided that more suitable characteristics could be obtained by operating as a pump at 120 per cent of the design speed, then the runaway speed would exceed that of normal operation by about 12 per cent.

Turbine Operation for Possible Peak-Load Power Development. The Grand Coulee pumping plant of course is only a part of the total Grand Coulee project. A major function of the latter is power development. One of the problems always confronting a power project is the provision of sufficient capacity to meet peak-load demands. Therefore, the possibility has been suggested of using the pumping plant as a peak-load power supply by allowing the water to flow back from the upper reservoir, thus operating the pumps as turbines and the synchronous motors as generators. It will be noted in both Figs. 14 and 15 that these units have zones of very high efficiency in the turbine region, practically identical with the maximum efficiency obtained as pumps. Since the power must be supplied at constant frequency, it is necessary that the speed of turbine operation be the same as that of the pump. It is, of course, desirable to get as much power as possible from the turbines. However, the zone of turbine operation is determined by the selection of the pump operating speed.

For example, if in Fig. 15, the pump is considered to operate at 100 per cent speed, the torque and therefore the horsepower available in the turbine region will be 75 per cent of the corresponding values for the pump. For the high-head condition, i.e., for 120 per cent head, the turbine output will go up to about 110 per cent of the normal pump input at 100 per cent head. If, however, a normal operating speed of 111 per cent is selected for the pump, as was shown to be desirable in Fig. 12, conditions are quite different. Now, it will be observed that the normal torque input to the pump is 130 per cent for the low-head condition and about 120 per cent for the high-head condition, whereas, the corresponding turbine operation shows a torque of only about 35 per cent for the low-head condition and about 85 per cent for maximum-head. These values must be corrected to the new reference of 130 per cent, which was the input torque to the pump under normal head conditions. On this basis, the turbine output varies from 27 to 60 per cent of the power input to the pump at normal operating head. This output would appear to be so small as to be of doubtful value for a peak-load power supply. The trend, indicated by these examples, appears to be general, i.e., for a given design, if the operating point as a pump is located at a relatively low capacity, the operating speed will be low, the

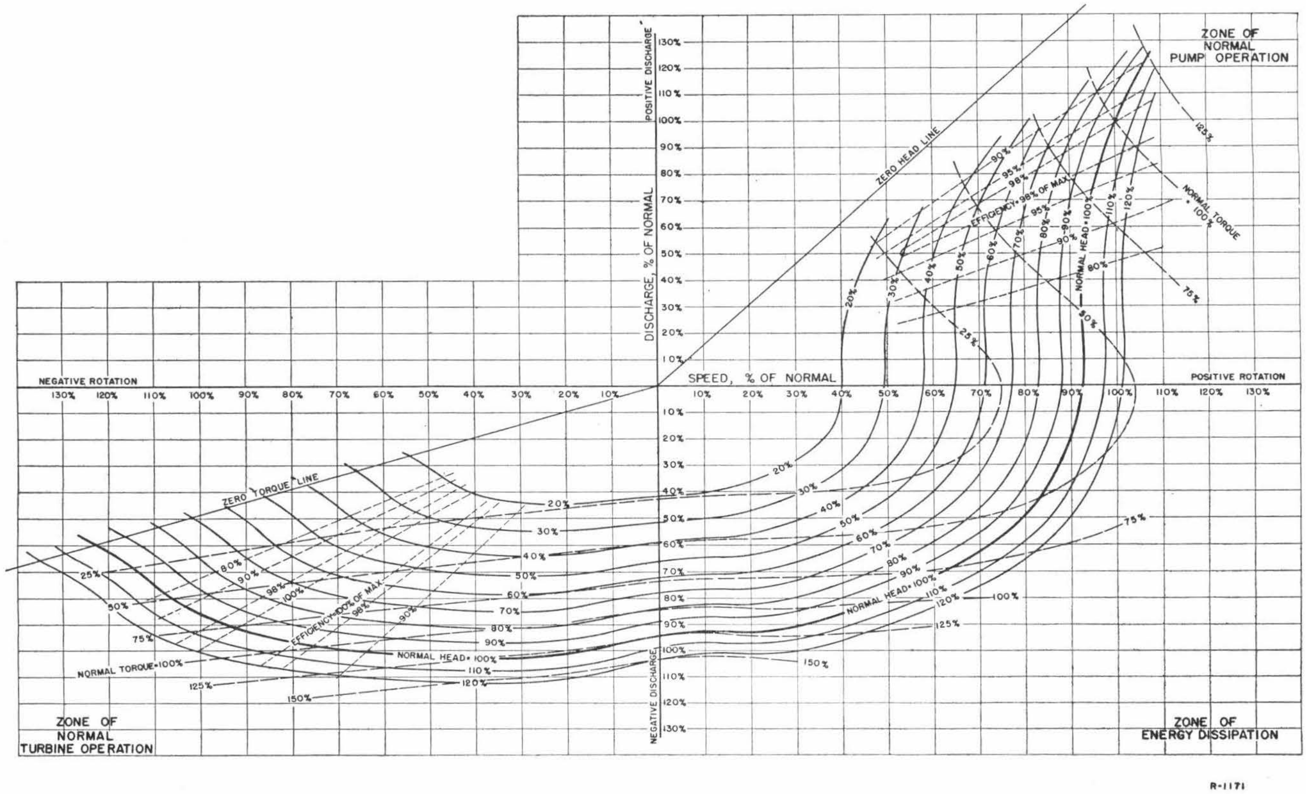

FIG. 14 COMPLETE CHARACTERISTIC DIAGRAM, SINGLE-VOLUTE PUMP, GROUP B

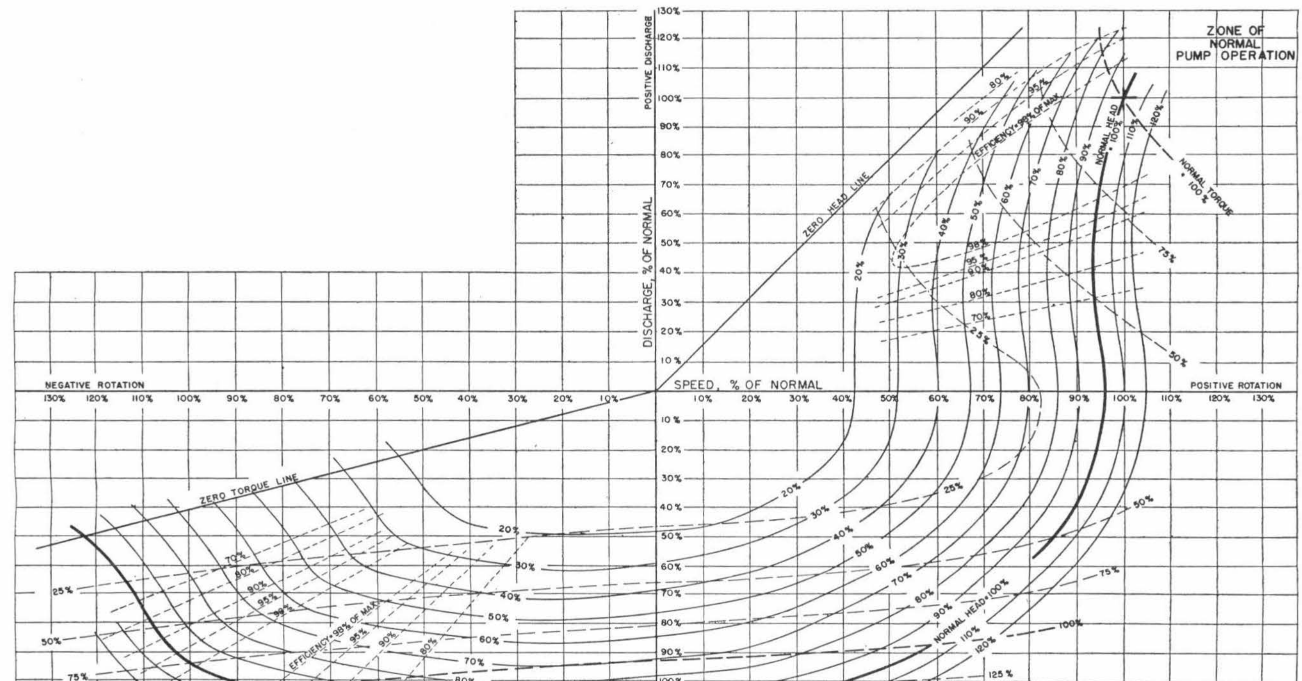

FIG. 15 COMPLETE CHARACTERISTIC DIAGRAM, DOUBLE-VOLUTE PUMP, GROUP B

turbine capacity will be high and the runaway speed will be high; whereas, if the operating point is chosen at a relatively high capacity and speed, the turbine capacity and the runaway speed will both be comparatively low. Thus, one more factor is added to the complicated set of requirements involved in the choice of the proper unit for the given installation.

Transient Behavior. The transient behavior of a pump is a function not only of the pump characteristics, but also of the pipe-line characteristics and other hydraulic and inertia features of the entire installation. The prediction of transient behavior has been briefly discussed in one of the previous references (4). Figs. 16 and 17 show typical transient characteristics for the double-volute pump of group B. These were computed by the

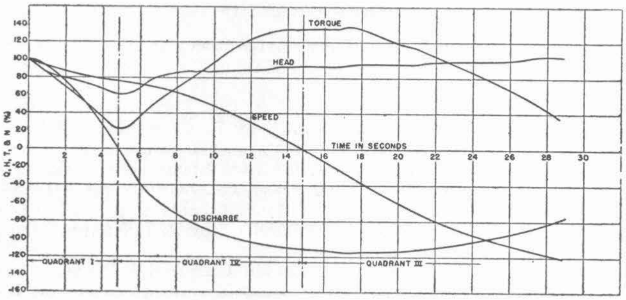

FIG. 16 TRANSIENT CHARACTERISTICS OF DOUBLE-VOLUTE PUMP, GROUP B

(Calculation for power failure when operating at 100 per cent speed and normal-head conditions.)

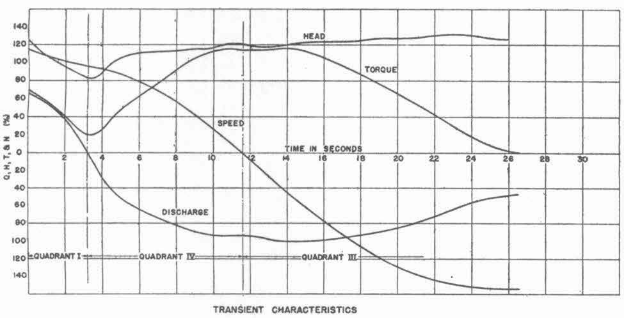

FIG. 17 TRANSIENT CHARACTERISTICS OF DOUBLE-VOLUTE PUMP, GROUP B

(Calculation for power failure when operating at 117.5 per cent speed and extreme high-head conditions.)

use of the Bergeron graphical method of water-hammer calculation (5) with the data from the laboratory for the complete pump characteristics.

Fig. 16 shows the performance following power failure when the pump has been operating at normal head and speed. Fig. 17 gives the corresponding characteristics for the extreme high-head condition, with the pump operating at 118 per cent of design speed at the time of power failure. For both conditions, it will be noted that the head fluctuations are quite moderate and present no problem. On the other hand, it is somewhat startling to imagine a 65,000-hp unit changing from a normal pump, operating at full speed in one direction, to a turbine operating at runaway speed in the other direction in an elapsed time of only 26 to 28 sec. The torque curves show that the maximum shaft stresses increase to 40 per cent above the normal operating value. The runaway speeds correspond closely to those already discussed.

Fig. 16 indicates that the unit operation remains in quadrant IV for as much as 10 sec. This is a region of complete energy dissipation, since energy is being poured into the machine through the deceleration of the rotating mass while, at the same time, energy is being given up in the machine by the fluid flowing through it. Little is known about cavitation conditions in this region, aside from the fact that they are apparently quite serious. It is felt that quadrant IV operation offers a fruitful field for further investigation.

SUMMARY OF RESULTS

Limitation of Program. Before summarizing the results, it should be re-emphasized that, although this investigation has shed some light on a few of the factors involved in the selection of the type and design of pump to meet particular needs of a given installation, the amount of information is still very meager. Many possibilities of casing design remain to be explored. Cavitation limits are yet too empirical in character, and the possibilities of obtaining more desirable performance for a given installation through changes in the impeller design are barely touched.

Operating Characteristics and Speed. The over-all performance of a pump, using a given impeller, is greatly affected by the case design. For a given type of case, the characteristics may be varied considerably by the choice of the point at which the case "fits" the impeller. Of the three types of cases studied, the double-volute type appears to give the widest high-efficiency range.

A well-designed impeller has a fairly wide range of speeds over which it will operate satisfactorily when delivering against a given head. A proper choice of case "fit" therefore will result in a unit having the desired operating speed. For a given combination of impeller and case, the head-capacity characteristics can be "steepened" by choosing the operating point at a relatively high capacity and speed. If a head-capacity steepness greater than that of the basic impeller performance is desired, it can be obtained only by the sacrifice of efficiency, i.e., by pushing the operating point to a capacity out beyond the zone of maximum efficiency. This is equivalent to using a casing too small for the desired capacity.

Hydraulic Balance and Radial Thrust. Within the zone of maximum efficiency, the fit of the case to the impeller is usually satisfactory enough to produce a relatively uniform pressure distribution around the impeller discharger. Therefore, operation in this zone is accompanied by little or no radial thrust. Operation at higher or lower capacities distorts this uniformity and results in radial thrust. The resultant force on the impeller and shaft is highest for the single-volute case. The fixed-vane-diffusor construction greatly reduces the magnitude of the force and it is eliminated by a well-designed double-volute casing.

Instability. Discontinuities in the head-capacity characteristic seem to be an inherent feature of centrifugal pumps, or at least of high-efficiency ones. These discontinuities probably are due to changes in the flow regimen, either in the impeller or case. They often limit the extent of the satisfactory operating range. The closeness with which they may be approached is presumably a function of the magnitude of the discontinuity.

Cavitation. Cavitation is an impeller phenomenon and is relatively insensitive to casing design. However, severe unbalance of the pressure distribution around the impeller discharge may change the cavitation conditions. Cavitation usually produces a change in the head-capacity characteristic. In general, the head is lowered, but under some circumstances it seems that it may be first increased. Cavitation forms one of the major limitations in determining the zone of satisfactory operation. If, in order to obtain other desirable characteristics, the operation point for a given impeller is chosen some distance away from the design point, it may be necessary to modify the eye design to secure satisfactory cavitation elimination. As yet, no satisfactory quantitative determination of the inception or degree of cavitation has been developed.

Turbine Operation. In general, a centrifugal pump can be operated very satisfactorily as a turbine and, over a limited range, with an efficiency equal to the best performance as a pump. In special cases, it may be feasible to utilize this possibility to supply a peak-load power demand by reversing the flow and operating the pump as a turbine and the motor as a generator. If this is to be done, careful consideration must be given to the design of the unit, since the selection of the pump operating point determines the turbine performance as well. The conditions for securing the optimum pump characteristics, turbine operation, and low runaway speed are usually not compatible, and therefore the relative value or the different elements of the performance must be evaluated carefully.

Conclusion

Although this study was designed to answer specific questions covering the selection of operating features for the pumping units to be installed at the Grand Coulee project, the results obtained are somewhat more generally applicable than might be expected. It is anticipated that, in the future, there will be more and more demand for hydraulic units, both pump and turbine, the characteristics of which are particularly adapted to the installation requirements, and it is felt that studies of the kind herein reported offer the most reliable means of securing the desired result.

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During the period of the investigations, the Bureau of Reclamation, through the chief engineer and members of his technical staff, has kept in intimate contact with the work and has contributed much to its progress. Especial acknowledgment is due to Mr. D. P. Barnes, resident representative, of the Bureau, who has taken an active part in the experimental investigations and their analyses.

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